

A Zonal Model For Predicting Simultaneous Heat And Moisture Transfer In Buildings

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ABSTRACT

To study the influence of adsorption/desorption by building materials on indoor air behavior, we have developed a zonal model to predict temperature and moisture fields in a room. This model is composed of two sub-models that represent indoor air conditions and room envelope. The heat and mass transfer across the building envelope are predicted by “evaporation and condensation” theory. The resulting set of non-linear coupled equations is solved simultaneously by the oriented-object platform, SPARK. In a study case with 27 zones the proposed model is compared to one without adsorption/desorption by building materials. The results indicate that adsorption/desorption influences the moisture field in confined spaces.

KEYWORDS

Building simulation, moisture transfer, zonal model, adsorption, SPARK.

INTRODUCTION

The importance of taking into account adsorption and desorption by building materials in building simulations has been emphasized in many studies ([KERES2], [WONG], [ELDIA]). Different models with different degrees of complexity are available in the literature to evaluate these phenomena ([KERES1], [ELDIA], [MENDE]). However, in whole-building simulations, these models are frequently coupled to indoor air models in which room conditions are considered uniform (lumped-parameter model). Therefore, the non-uniform behavior of the air inside the room, which is important in comfort analysis, cannot be evaluated.

Zonal models ([INARD], [MUSY]) are an alternative to CFD. In zonal models a room is divided into a relatively small number of zones—typically on the order of tens to hundreds—compared to thousands and more for typical CFD simulations. As a result zonal models execute much faster than CFD calculations. While not as fine-grained as CFD

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simulation, zonal models do give useful information about temperature and moisture distributions that is not available from lumped-parameter models.

In this work, the zonal model is composed of two sub-models that represent indoor air conditions and room envelope. In the indoor air sub-model, each zone into which the room is divided is described by a set of non-linear equations consisting of mass and energy balance equations and state equations. Convection and diffusion phenomena are taken into account in the evaluation of heat and moisture exchanges due to the interaction among zones and among zones and the outdoor environment. The heat and mass transfer across the building envelope are calculated by “evaporation and condensation” theory ([KERES1]). The resulting set of non-linear coupled equations is solved simultaneously by the object-oriented simulation environment, SPARK ([SOWEL]).

To evaluate the influence of adsorption/desorption by building materials on indoor air behavior, we compared the proposed model to one that doesn't consider adsorption and desorption phenomena. This latter model is the same as the proposed model with the exception that the heat transfer across the envelope is calculated with the well-known conduction equation neglecting moisture effects.

MODEL DESCRIPTION

In the indoor air sub-model each zone is characterized by two basic sets of equations. The first set is composed of mass and energy balance equations and state equations that determine the physical characteristics of the moist air in the zone. The second set is composed of equations that determine the heat and mass exchange between the zone and its neighboring zones.

Assuming that the moist air in a zone is well mixed, the mass (dry air and water vapor) and energy balance equations for a zone can be written as follows:

$$\sum_{i=1}^6 \dot{m}_{da} = 0 \quad (1)$$

$$V \frac{d\rho_{wv}}{d\tau} = \sum_{i=1}^6 \dot{m}_{wv} + \dot{m}_{wv,source} \quad (2)$$

$$\rho_{da} C_{p_{da}} V \frac{dt_{ma}}{d\tau} = \dot{q}_{source} + \sum_{i=1}^6 \dot{Q} + \sum_{i=1}^6 \phi \quad (3)$$

where

$C_{p_{da}}$	= specific heat of dry air at constant pressure (J/kg°C)
i	refers to face i of the zone
\dot{m}_{da}	= dry air mass flow rate (kg/s)
\dot{m}_{wv}	= water vapor mass flow rate (kg/s)
$\dot{m}_{wv,source}$	= water vapor source (kg/s)
\dot{q}_{source}	= sensible heat source (W)
\dot{Q}	= time rate of heat transfer transported by dry air mass flow (W)
t_{ma}	= temperature of moist air (°C)
V	= volume (m ³)
ρ_{da}	= dry air density (kg/ m ³)
ρ_{wv}	= water vapor density(kg/ m ³)
ϕ	= time rate of conductive heat transfer (W)
τ	= time (s)

Note in Eq. (1) that the temporal variation of dry air mass inside the zone has been neglected.

The physical characteristics of the zone are coupled by the perfect gas law, Eq. (4), and by psychometric relations, Eqs. (5) – (7):

$$P_{ma} = (\rho_{da} \mathcal{R}_{da} + \rho_{wv} \mathcal{R}_{wv}) T_{ma} \quad (4)$$

$$\rho_{ma} = \rho_{da} + \rho_{wv} \quad (5)$$

$$W = \rho_{wv} / \rho_{da} \quad (6)$$

$$\phi = (\rho_{wv} \mathcal{R}_{wv} T_{ma} / P_{wv,sat})_{P_{ma}, T_{ma}} \quad (7)$$

where

P_{ma}	= total pressure of the zone (Pa)
$P_{wv,sat}$	= total vapor pressure of water in saturated moist air (Pa)
T_{ma}	= temperature of moist air (K)
W	= humidity ratio of moist air
ρ_{ma}	= moist air density (kg/ m ³)
ϕ	= relative humidity
\mathcal{R}_{da}	= dry-air constant (J/kgK)
\mathcal{R}_{wv}	= water vapor constant (J/kgK)

One additional equation is used to calculate the water vapor pressure in equation (7). In our study, we adopt that one proposed by Woloszyn [WOLOS].

The inter-zonal air movement due to the pressure difference between two neighboring zones is based on the orifice flow equation. This flow is calculated differently for vertical and horizontal interfaces because hydrostatic effects are taken into account in the latter. For vertical interfaces the dry-air mass flow rate can be calculated as follows:

$$p_{ma,1} - p_{ma,2} \geq 0 \quad \dot{m}_{da} = CdA \rho_{da,1} \sqrt{2(p_{ma,1} - p_{ma,2}) / \rho_{ma,1}} \quad (8a)$$

$$p_{ma,1} - p_{ma,2} < 0 \quad \dot{m}_{da} = -CdA \rho_{da,2} \sqrt{2(p_{ma,2} - p_{ma,1}) / \rho_{ma,2}} \quad (8b)$$

where

A	= area (m ²)
Cd	= discharge coefficient
p_{ma}	= relative pressure of the zone (Pa)

The indices 1 and 2 correspond, respectively, to the left and right zone separated by the vertical interface.

The same equations can be used in the case of a horizontal interface, except that in this case the pressure at the interface (p_{ma}^*) depends on zone dimensions:

$$p_{ma,1}^* = p_{ma,1} + \rho_{ma,1} g(h_1/2) \quad (9a)$$

$$p_{ma,2}^* = p_{ma,2} - \rho_{ma,2} g(h_2/2) \quad (9b)$$

where g is the gravitational acceleration (m/s²), h is the height of the zone (m), and indices 1 and 2 correspond, respectively, to the zone above and below the interface.

For both vertical and horizontal interfaces, the water vapor mass flow is composed of two terms: a diffusive term and a term due to the pressure difference between neighboring zones. The water vapor mass flow due to the pressure difference is based on the dry-air mass flow, as given by Eqs. (10a) and (10b), whereas the diffusive term is calculated by Fick's law, Eq. (11):

$$p_{ma,1} - p_{ma,2} \geq 0 \quad \dot{m}_{wv} = \rho_{wv,1} (\dot{m}_{da} / \rho_{da,1}) \quad (10a)$$

$$p_{ma,1} - p_{ma,2} < 0 \quad \dot{m}_{wv} = \rho_{wv,2} (\dot{m}_{da} / \rho_{da,2}) \quad (10b)$$

$$\dot{m}_{wv} = D_{wv} A [(\rho_{ma,1} + \rho_{ma,2})/2] \frac{(\rho_{wv,1}/\rho_{ma,1} - \rho_{wv,2}/\rho_{ma,2})}{[(l_1 + l_2)/2]} \quad (11)$$

where D_{wv} is the coefficient of water vapor diffusion in air and l is the zone's dimension perpendicular to the interface (m).

The overall heat exchange across an interface is the sum of the enthalpy flux given by Eqs. (12a) and (12b), and a conductive flux given by equation (13).

$$\dot{m}_{da} \geq 0 \quad \phi_t = \dot{m}_{da} C p_{da} t_{ma,1} \quad (12a)$$

$$\dot{m}_{da} < 0 \quad \phi_t = \dot{m}_{da} C p_{da} t_{ma,2} \quad (12b)$$

$$\dot{Q} = kA \frac{(t_{ma,1} - t_{ma,2})}{(l_1 + l_2)/2} \quad (13)$$

where k is the thermal conductivity of moist air (W/mK).

The heat and mass transfer across the building envelope is expressed by “evaporation and condensation” theory (Kerestecioglu and Gu, 1990), which is valid in the materials' hygroscopic range. In other words, moisture is supposed to migrate through the solid material in its vapor phase. This model is basically composed of a water balance equation, an energy balance equation and an adsorption isotherm. In our work, this set of coupled equations is solved using the “finite difference” method. This sub-model is applied at each zone face that is adjacent to the room envelope.

RESULTS

The proposed model and its variant that ignores adsorption/desorption phenomena were compared in a study case composed of 27 zones. This study case represents a room with an opening at the top of the West side and another at the bottom of the opposite side.

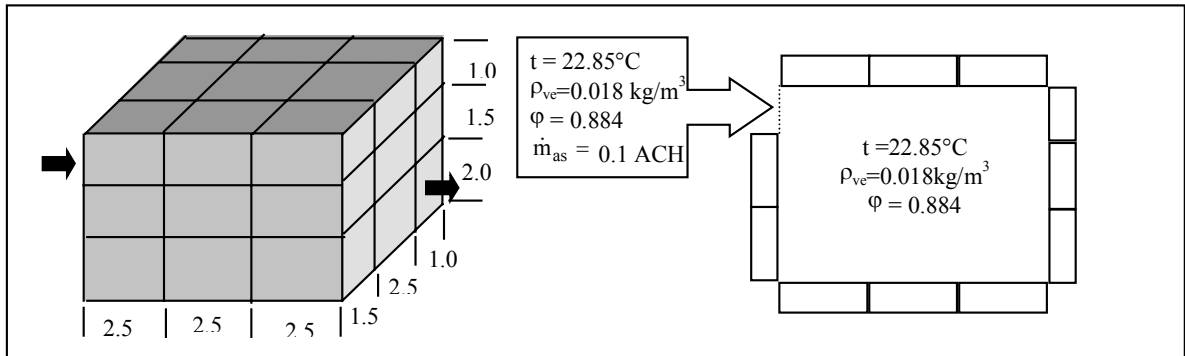


Figure 1: Sketch of the studied room.

Figure 1 shows a sketch of this room with the zones' dimensions (m), the initial conditions inside the room, and the conditions of the air entering through the West side opening. The discharge coefficient and the coefficient of water vapor diffusion in the air adopted in the interfaces and openings are 0.37 and $0.00255 \text{ m}^2/\text{s}$, respectively. We consider that the room envelope is constituted of a high hygroscopic material, initially at $t = 26.85^\circ\text{C}$ and $\rho_{ve} = 0.013 \text{ kg/m}^3$ ($\phi = 0.50$). The exterior walls are surrounded by air at the same conditions of the room envelope. The convective heat and

mass transfer coefficients are respectively $5.0 \text{ W/m}^2\text{°C}$ and 0.005 m/s , for both external and internal surfaces. External and internal radiation exchanges are not taken into account.

Figure 2 shows the calculated temperature distribution vs. time in the three zones in the middle of the room for the models with and without moisture adsorption/desorption. The temperature fields predicted by both models are similar. Additionally, in both cases the temperature of the highest zone increases more than the temperature of the other zones.

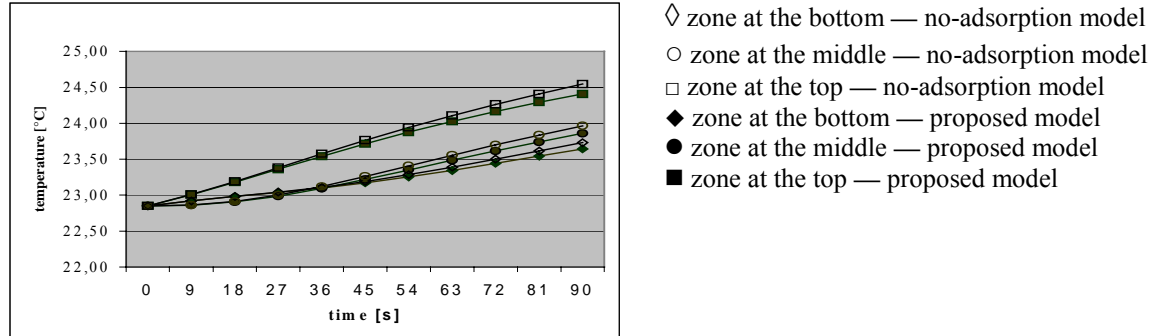


Figure 2: Calculated temperature distribution in the three zones in the middle of the room as predicted by zonal models with and without moisture adsorption/desorption.

Figure 3 shows the corresponding water vapor density distribution in the three zones in the middle of the room for the two models. Unlike the temperature distribution results, the two models give significantly different behavior. The water vapor density is almost constant when adsorption/desorption is not considered; otherwise it decreases significantly with time. We also note that the decrease with time is largest for the highest zone.

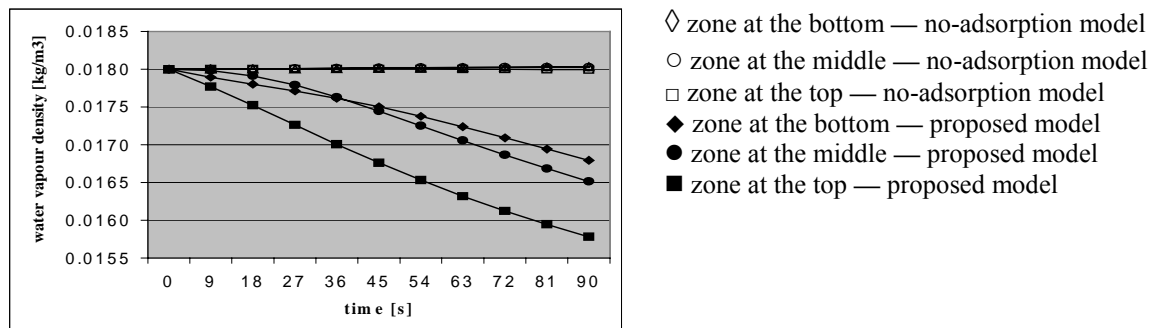


Figure 3: Water vapor density distribution vs. time in the zones in the middle of the room space as predicted by zonal models with and without moisture adsorption/desorption.

We attribute the differences found in the temperature and water vapor density in the highest zone to the weak airflow across it.

DISCUSSION

Despite the fact that the results represent a short simulation time, they indicate that adsorption and desorption phenomena influence the indoor air behavior. The reduced time of simulation is due to convergence problems presented by the proposed model. We note that the number of equations reduces significantly when the conduction equation is used to represent the room envelope (from 1612 to 832 in the envelope sub-model).

Simple models that represent the heat and mass transfer across the building envelope, such as the “Effective Moisture Penetration Depth – EMPD” theory ([KERES2]) can be an alternative to reduce the number of equation in the envelope sub-model. However, since such a model considers that only a thin surface layer of the material is affected by the indoor air conditions, it is limited to short time periods (on the order of a few hours).

CONCLUSION AND IMPLICATIONS

A dynamic model for predicting moist air behavior in internal spaces using the zonal approach has been proposed. Due to the zonal aspect of the model, both temperature and moisture fields can be calculated, which is not possible with single-node models. The comparison of this model with one that ignores moisture adsorption/desorption shows that moisture adsorption/desorption influences the moisture field inside the room.

Besides evaluating the influence of adsorption and desorption phenomena on indoor air conditions, the zonal model allows evaluation of the influence of other parameters, such as internal heat gain, moisture sources, and outside air conditions.

We are currently working on integrating a radiant exchange model to complete the global model. We are also working on a cooling system model that will allow thermal comfort in conditioned spaces to be evaluated. Additionally, it will allow studying how to locate a control sensor inside a room so as to reduce cooling energy while maintaining occupant comfort.

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